

Variable Speed Drive (VSD) Applications in Dual-Duct Constant Volume Systems

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ABSTRACT

Models have been developed for static pressure and potential supply fan energy savings by using variable speed drive (VSD) in dual-duct constant volume systems. Experiments have been performed using a full size dual-duct constant volume system installed in a 68,000 ft² (6,317 m²) office and classroom building. The measured static pressure variations and the energy savings agree with the model projected values. The VSD saves the fan power by as much as 35%, reduces the total airflow by 15%, and decreases the excessive static pressure on the terminal box dampers. This paper presents the systems models, the experimental methods and the results.

INTRODUCTION

Single-fan, dual-duct (SFDD) constant volume air-handling units (AHU) have been installed in many medical facilities, office buildings and library facilities since 1940's. They are especially popular in hot and humid climates because they offer good temperature and humidity control. When a constant speed fan is used, the static pressures in both the hot and cold ducts are higher than the design specifications under partial load conditions. Terminal box dampers are often over-pressurized, which creates noise and vibration problems and causes excessive airflow in parts of the building where the single actuator terminal boxes are used [Liu et al., 1997].

The performance of the SFDD systems can be improved by converting them to dual-fan, dual-duct (DFDD) systems [Joo and Liu, 2002]. In such a conversion a dedicated hot air fan is added to the system, and variable speed drives are added to both the hot and cold air fans. The fan speeds are controlled to maintain the required static pressures at the selected duct locations. The dual-fan conversion requires major mechanical retrofits.

This paper investigates the potential supply air fan energy savings by using VSD in single-fan, dual-

duct constant volume systems. The VSD modulates the fan speed to maintain the lower of the hot duct and cold duct static pressures at the set point. The theoretical models have been developed. Experiments have also been performed using a full size system in a 68,000 ft² building. Both the theoretical models and experiments are presented in this paper.

MODELS

The single-fan, dual-duct constant volume unit provides both hot air and cold air to rooms where terminal boxes mix cold and hot air to accommodate the room load variation. When the cooling load decreases, the hot air flow increases and the cold airflow decreases. When the cooling load increases, the hot air flow decreases and the cold airflow increases. The total airflow of the AHU remains constant.

Figure 1 presents a single-fan, dual-duct constant volume system with static pressure control. The static pressure control system consists of a controller, two static pressure sensors and a variable speed drive.

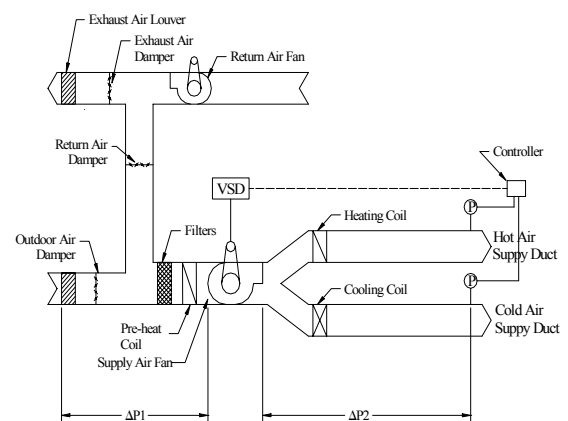


Figure 1. Schematic diagram of a dual-duct system with static pressure control

One static pressure sensor is located in the cold duct, and the other is located in the hot duct. It is suggested to locate the sensors in the main ducts where the nearest terminal box is attached. The static pressure set point should be high enough to deliver the required airflows to each terminal box. The controller selects the lower static pressure value from the two sensors and compares it with the set point. If the measured value is lower than the set point, the variable speed device speeds up the fan. If the measured value is higher than the set point, the variable speed device slows down the fan speed. The static pressure control system maintains the lower static pressure at the set point.

Constant Speed Fan Systems (CSFS)

The constant speed fan system is selected as the base system. Its models are developed here first. The pressure losses in the dual-duct constant volume systems consist of three parts: (1) inlet pressure loss (ΔP_1) from outside to the fan inlet, (2) main duct pressure loss (ΔP_2) from the fan discharge to the static pressure sensors, and (3) downstream pressure loss (ΔP_3) from the static pressure sensors to the building space. It is assumed that the room pressure equals the outside pressure. Under the design condition, 100% air flows through the cold air duct. During this condition, the main duct pressure loss reaches the maximum value, while the downstream pressure loss reaches the minimum value or the design value. The fan pressure head equals the sum of the pressure losses:

$$H_d = \Delta P_{1,d} + \Delta P_{2,d} + P_{st,d} \quad (1)$$

Equation (1) is rewritten as a dimensionless format by dividing both sides by H_d .

$$1 = \chi_1 + \chi_2 + \chi_3 \quad (2)$$

$$\text{Where: } \chi_1 = \Delta P_{1,d} / H_d, \chi_2 = \Delta P_{2,d} / H_d, \\ \chi_3 = P_{st,d} / H_d$$

The inlet pressure loss (ΔP_1) is assumed to be constant and considered as a fraction (α) of the design static pressure at main ducts ($P_{st,d}$).

$$\chi_1 = \alpha \cdot \chi_3 \quad (3)$$

Inserting equation (3) into Equation (2), the static pressure fraction becomes:

$$\chi_3 = \frac{1}{1 + \alpha} (1 - \chi_2) \quad (4)$$

Under partial load conditions, the fan head remains constant if the total airflow is assumed to be constant. The pressure loss from outside to the fan inlet remains unchanged. However, the pressure losses from fan discharge to the static pressure sensors vary proportionally to the square of the hot or cold airflow. The decreased pressure losses in the main ducts are consumed by the terminal box dampers.

$$H_d = \Delta P_{1,d} + \Delta P_{2,d} \cdot \bar{Q}_{\min}^2 + P_{st,\max} \quad (5a)$$

$$H_d = \Delta P_{1,d} + \Delta P_{2,d} \cdot \bar{Q}_{\max}^2 + P_{st,\min} \quad (5b)$$

$$\text{Where: } \bar{Q}_{\min} = \min(Q_c / Q_{c,\max}, Q_h / Q_{h,\max}), \\ \bar{Q}_{\max} = \max(Q_c / Q_{c,\max}, Q_h / Q_{h,\max})$$

Pressure losses are again represented as fractions:

$$1 = \chi_1 + \chi_2 \cdot \bar{Q}_{\min}^2 + \chi_3 \cdot \beta_{\max} \quad (6a)$$

$$1 = \chi_1 + \chi_2 \cdot \bar{Q}_{\max}^2 + \chi_3 \cdot \beta_{\min} \quad (6b)$$

$$\text{Where: } \beta_{\max} = P_{st,\max} / P_{st,d}, \\ \beta_{\min} = P_{st,\min} / P_{st,d}$$

The static pressure ratio (β) is referred to as the ratio of the actual static pressure over the design static pressure. β_{\max} and β_{\min} are the larger and smaller of the hot and cold duct static pressure ratios, respectively.

Inserting equation (3) into equations (6a) and (6b) yields:

$$\beta_{\max} = 1 + \frac{\chi_2 (1 + \alpha) (1 - \bar{Q}_{\min}^2)}{1 - \chi_2} \quad (7a)$$

$$\beta_{\min} = 1 + \frac{\chi_2 (1 + \alpha) (1 - \bar{Q}_{\max}^2)}{1 - \chi_2} \quad (7b)$$

Figure 2 shows the simulated maximum and minimum static pressure ratios (β_{\max} and β_{\min}) of the CSFS depending on the cold airflow ratio for

different main duct pressure loss fractions ($\chi_2 = 0.3 \sim 0.7$). The inlet pressure loss is assumed to be the same as the design static pressure ($\alpha = 1$). Since the size of hot duct is generally designed smaller than the cold duct, the design hot airflow rate must be smaller than the design cold airflow rate at the same design fan head. In the calculation, the design hot airflow rate is assumed to be 70% of the design cold airflow rate.

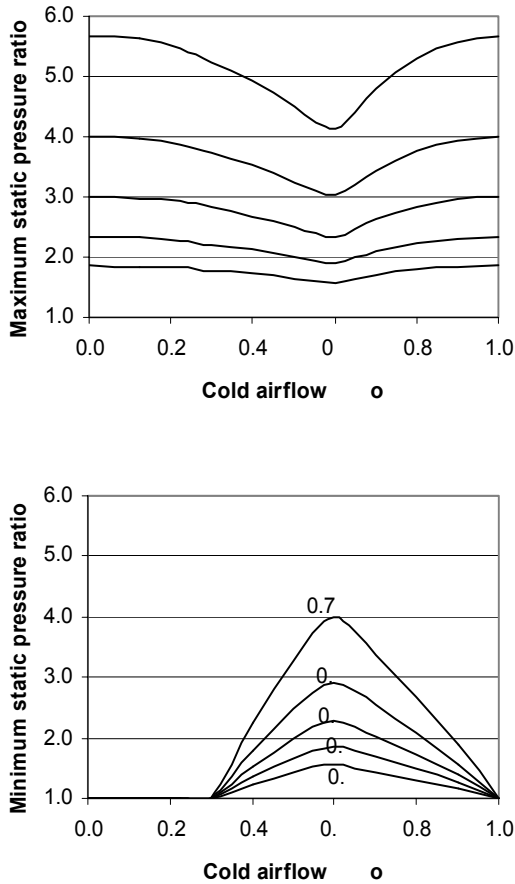


Figure 2. Maximum and minimum static pressure ratio vs. cold airflow ratio for the CSFS

The maximum static pressure is always higher than the design value (Figure 2, top). The minimum static pressure is also higher than the design value except in 100% cooling or 100% heating conditions (Figure 2, bottom). The higher the main duct static pressure fraction, the higher the maximum and minimum static pressures are. The unnecessary high static pressures in both the cold and hot ducts indicate fan power wastes, possible airflow control and noise problems in terminal boxes.

Variable Speed Fan Systems (VSFS)

The design fan head for the dual-duct constant volume (DDCV) systems with the VSFS consists of the same pressure loss and static pressure as the DDCV systems with CSFS.

$$H_d = \Delta P_{1,d} + \Delta P_{2,d} + P_{st,d} \quad (8)$$

A variable speed device modulates the fan speed to maintain actual static pressure at the design static pressure under partial load conditions.

$$H_v = \Delta P_1 + \Delta P_{2,d} \cdot \bar{Q}_{\max}^2 + P_{st,d} \quad (9)$$

Since both hot and cold branches have the same start location (outside) and the same end location (room), the hot duct branch pressure loss equals the cold duct branch pressure loss.

$$\Delta P_1 + \Delta P_{2,d} \cdot \bar{Q}_c^2 + P_{st,c} = \Delta P_1 + \Delta P_{2,d} \cdot \bar{Q}_h^2 + P_{st,h} \quad (10)$$

$$\text{Where: } \bar{Q}_c = Q_c / Q_{c,\max}, \bar{Q}_h = Q_h / Q_{h,\max}$$

If the cold airflow ratio (\bar{Q}_c) is higher than the hot airflow ratio (\bar{Q}_h), the static pressure in the cold air duct is the same as the set point. If the cold airflow ratio is smaller than the hot airflow ratio, the static pressure in the cold air duct is higher than the set point, and vice versa.

$$\Delta P_{2,d} \cdot \bar{Q}_{\min}^2 + P_{st,\max} = \Delta P_{2,d} \cdot \bar{Q}_{\max}^2 + P_{st,d} \quad (11)$$

Equation (11) can be rewritten as:

$$\beta_{\max,v} = \frac{P_{st,\max}}{P_{st,d}} = 1 + \frac{\Delta P_{2,d}}{P_{st,d}} (\bar{Q}_{\max}^2 - \bar{Q}_{\min}^2) \quad (12)$$

Therefore, the maximum static pressure ratio for the VSFS becomes a function of the main duct pressure loss fraction (χ_2), the equipment pressure loss fraction (α) and the maximum and minimum airflow ratios in the hot duct and the cold duct.

$$\beta_{\max,v} = 1 + \frac{\chi_2(1+\alpha)(\bar{Q}_{\max}^2 - \bar{Q}_{\min}^2)}{1 - \chi_2} \quad (13)$$

Figure 3 presents the simulated results of the maximum static pressure ratio ($\beta_{\max,v}$) versus cold airflow ratio for the VSFS ($\chi_2 = 0.3 \sim 0.7$). The assumptions for the inlet pressure loss and the ratio of the design hot and cold airflow are the same as those in the CSFS.

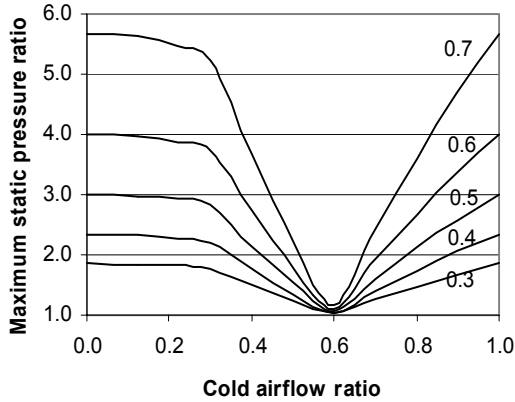


Figure 3. Maximum static pressure ratio vs. cold airflow ratio for the VSFS

The VSFS maintains the minimum static pressure at the design set point regardless of the airflow ratios. The VSFS controls the maximum static pressure to the design value when the hot airflow ratio equals the cold airflow ratio. The VSFS has much lower maximum static pressure than the CVFS except under the full load conditions. The reduction of the maximum static pressure ratio equals the difference of the maximum and minimum static pressures of the CSFS. The lowered maximum static pressure ratio indicates the potential fan energy savings and less airflow control and noise problems in terminal boxes.

Power Savings

The fan power savings is defined as the difference between the fan powers of CFVS and VSFS systems. Assuming the same fan efficiency for both the CSFS and the VSFS systems, the fan power savings ratio is expressed by equation (14).

$$\phi_f = 1 - \frac{E_v}{E_d} = 1 - \frac{\Delta P_{1,d} + \Delta P_{2,d} \cdot \bar{Q}_{\max}^2 + P_{st,d}}{(1 + \delta)(\Delta P_{1,d} + \Delta P_{2,d} + P_{st,d})} \quad (14)$$

The fan power savings ratio can be rewritten as:

$$\phi_f = \frac{\delta + \chi_2(1 - \bar{Q}_{\max}^2)}{1 + \delta} \quad (15)$$

The correction factor (δ) takes into the account of the extra airflow of the CSFS systems. Due to excessive static pressure on the terminal box dampers, the actual airflow may be higher than the design value.

Figure 4 presents the potential fan power savings of the VSFS ($\chi_2 = 0.3 \sim 0.7$). Under the design conditions, the fan power consumption of the VSFS is the same as that of the CSFS. Under partial load conditions while the air is distributed through both hot and cold ducts, the pressure loss of each main duct is significantly lower than the design value. This creates the fan power savings opportunities for the VSFS. The higher the main pressure loss, the higher the fan power savings is. Since the cold airflow ratio usually varies from 20% to 80%, the savings range from 10% to 40% around the year.

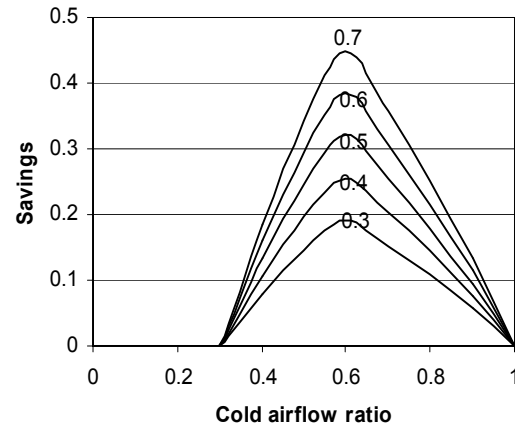


Figure 4. Fan power savings vs. cold airflow ratio (δ is assumed to be zero)

EXPERIMENTAL VERIFICATION

Experiments are conducted in an existing dual-duct constant volume system that serves a four-story building. The objective of the experiment is to verify the static pressure ratio and the fan power savings projected by the theoretical models. The pressure loss fractions are determined from field measurements, AHU's design and operational information. The maximum and the minimum pressure ratios and the fan power saving ratio are simulated by using the building and AHU information, and the simulated results are compared with the measured values.

Facility

The experiments were conducted in a full-size single-fan, dual-duct constant air volume system, which serves a four story building with a gross floor area of 68,000 ft² (6,317 m²). The unit was installed in 1960s. The initial design airflow rate was 57,000 ft³/min (26.9 m³/s) supplied with a 100 hp (74.6 kW) fan. The AHU is located in the attic. Currently, the motor is downsized to 60 hp (44.8 kW) to avoid the noise problem. The total airflow rate is 48,000 ft³/min (22.7 m³/s).

The hot duct size is smaller than the cold duct size for the same routes. For the same fan head, the design hot airflow rate is smaller than the design cold airflow rate that is usually used for selecting the fan size. By calculating the design main duct pressure loss from the fan discharge to the static pressure sensor in each duct, the design hot airflow rate is determined as 31% of the design cold airflow rate.

Field tests measured the fan head of 3.5 inH₂O (872 Pa) at full speed. The total airflow rate was 48,300 ft³/min (22.8 m³/s). The static pressure reading was 1.45 inH₂O (361 Pa). The cold airflow was determined as 76% of the total airflow. If 100% air flows through the cold air duct (design condition), the main duct pressure loss ($\Delta P_{2,d}$) is 2.2 inH₂O (548 Pa). The design static pressure ($P_{st,d}$) is 0.5 inH₂O (125 Pa). The inlet pressure loss ($\Delta P_{1,d}$) is 0.8 inH₂O (199 Pa). Therefore, the main duct pressure loss fraction (χ_2) is 0.6286.

Instrumentation

The EMCS (energy management and control system) of the campus facility was used to collect the following hourly data: the cold and hot duct static pressures at the sensor locations, the total airflow rates, the supply fan differential pressure, the cooling and heating energy consumptions, and the fan power. Prior to the data collection, all sensors are checked using hand-held meters.

The total airflow was measured by vortex-shedding meters installed in the suction side of the fan. A total of 2 sensors are used. The hot and cold airflow rates were calculated from the cooling and heating energy consumptions, the measured mixed air temperature and enthalpy, and the cold air and hot air temperatures and enthalpies. The cooling energy was measured by a vortex-shedding flow meter and two temperature sensors. The condensate flow was measured by a rotary drum condensate meter. The

heating energy was then determined based on a constant latent heat rate (1,000 Btu/lbm or 2,326 kJ/kg). The fan power is measured using a true power meter, which measures both the KVA and the power factor. The fan power is determined as the product of the KVA and the power factor.

The data from February 11th, 2001 to February 27th, 2001 were used for the analysis. The supply fan was controlled by a variable frequency drive for the last 8 days, and it was operated at its full speed for the first 8 days.

For the VSFS experiment, the static pressure set point of 0.7 inH₂O (174 Pa) was used.

Results and Discussions

Figure 5 presents the measured hourly total airflow under both the CSFS and the VSFS operations. The airflow is decreased from 48,000 ft³/min (22.7 m³/s) to 41,000 ft³/min (19.3 m³/s) when the operation is switched from the CSFS to the VSFS. The excessive airflow factor (δ) is 0.178 in this case. The airflow reduction indicates that the terminal boxes are actually pressure dependent. This issue will be discussed in another paper.

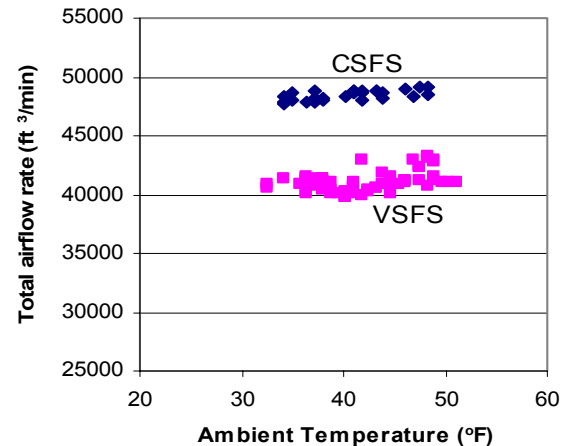


Figure 5. Measured hourly total airflow for both the CSFS and the VSFS operations

Figure 6 compares the theoretical and the measured maximum and minimum static pressure ratios (β_{\max} and β_{\min}) when the fan is at the constant speed. The center line shows the theoretical values, and the dots show the measured data. The broken lines shows error ranges based on the specification of sensors. The procedure of error analysis is explained in Appendix. The measured maximum and minimum static pressure ratios fit the

theoretical model within the expected error ranges. However, the large variation could indicate that more accurate measurements are needed to reduce the scatter.

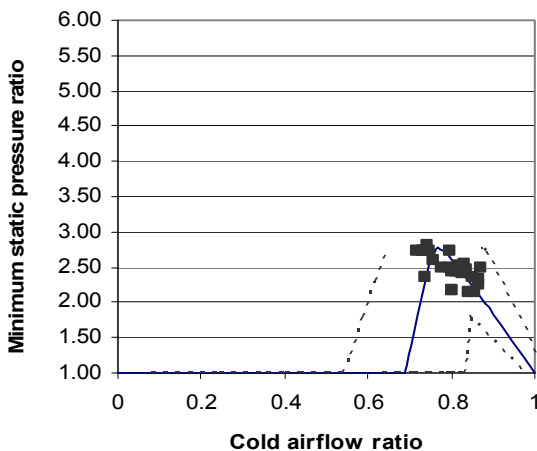
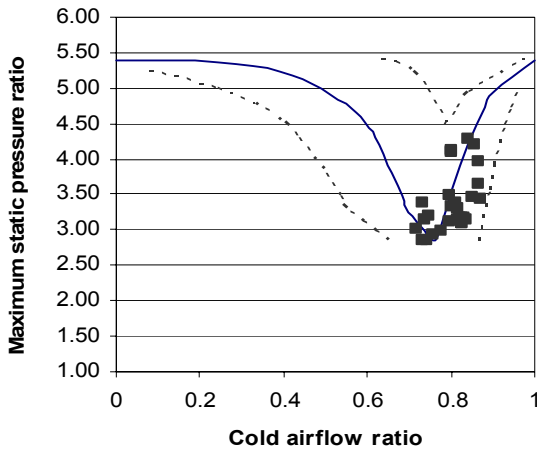


Figure 6. Comparisons of the simulated and measured results of maximum and minimum static pressure ratios vs. cold airflow ratio for the CSFS

Figure 7 compares the theoretical and the measured maximum static pressure ratio ($\beta_{\max,e}$) versus the cold airflow ratio for the VSFS. The measured data match the theoretical model. The maximum static pressure ratio ranges from 1.0 to 2.5 (0.7 inH²O (174 Pa) ~ 1.75 inH²O (436 Pa)) during the operating period. Compared with average maximum static pressure ratio of 3.4 (1.7 inH²O (423 Pa)) for the CSFS, the maximum static pressure is 1.6 (1.1 inH²O (274 Pa)) for the VSFS. The amount of reduction yields the fan power savings.

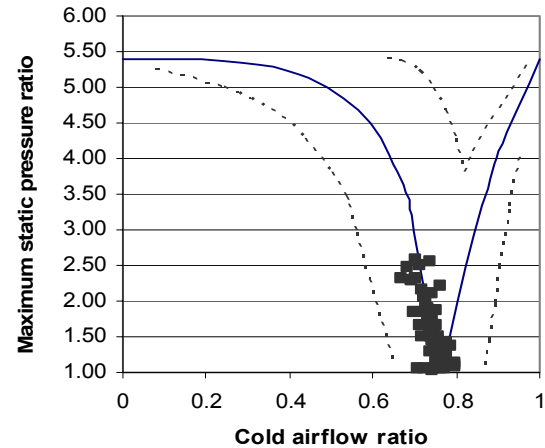


Figure 7. Comparisons of the simulated and measured results of maximum static pressure ratio vs. cold airflow ratio for the VSFS

Figure 8 compares the measured hourly supply fan power of the CSFS and VSFS systems. The average fan power was 35.8 kW for the CSFS operation. The average fan power is 23.1 kW for the VSFS operation. The average fan power savings is 12.7 kW or 35%.

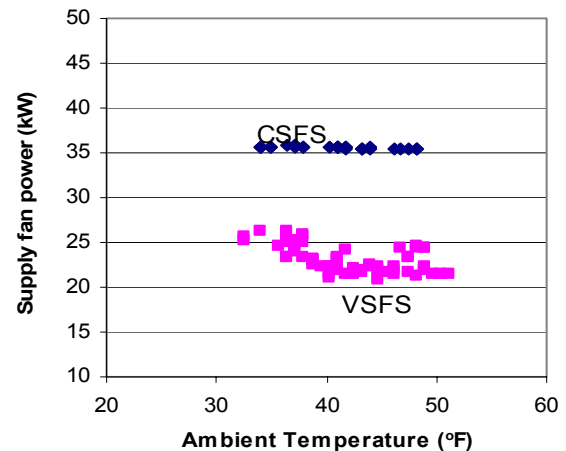


Figure 8. Measured hourly supply air fan power for both the CSFS and VSFS operations

Figure 9 compares the theoretical and the measured fan power energy savings. The fan power savings ranges from 30% to 40%. Assuming the savings rate of 35%, the annual fan power savings would be 109,763kWh. When the electricity costs \$0.05/kWh, about \$5,488 can be saved annually for simply installing a 60 hp VFD to the dual-duct constant air volume system in this building.

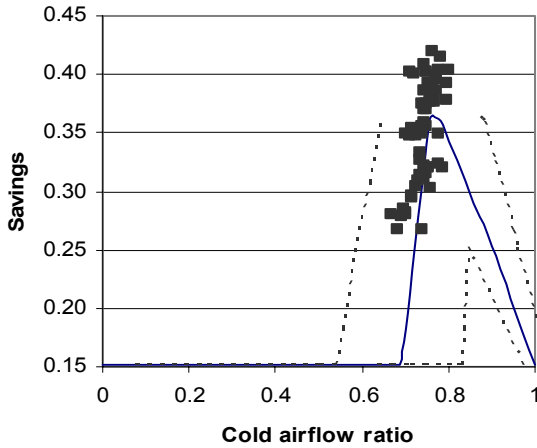


Figure 9. Comparison of the simulated and measured results of fan power energy savings vs. cold airflow ratio by converting from the CSFS to the VSFS

CONCLUSION

Analytical models have been developed for both the conventional single-fan, dual-duct constant air volume system and the variable speed fan system (VSFS). The fan power energy savings of the VSFS is simulated for different load conditions and system configurations. The energy performance of the VSFS was tested in a full-size AHU.

The measured results agree with the theoretical values. The VSFS reduces the maximum static pressure by setting the minimum static pressure at the set point, which leads to a substantial amount of fan power savings compared to usual operation conditions. The savings were average 35% of the original fan power consumptions.

NOMENCLATURE

E	= Fan power energy consumption (kW)
H	= Design fan head (Pa or inH ₂ O)
P_{st}	= Static pressure (Pa or inH ₂ O)
ΔP_1	= Pressure loss from outside to fan inlet (Pa or inH ₂ O)
ΔP_2	= Pressure loss from fan discharge to static pressure sensors (Pa or inH ₂ O)
ΔP_3	= Pressure loss from static pressure sensors to building space (Pa or inH ₂ O)
Q	= Airflow rate (kg/s or lbm/hr)
\overline{Q}	= Airflow ratio (Q/Q_{\max})
α	= Fraction ($\Delta P_1 / P_{st,d}$)
β	= Ratio of actual static pressure over design static pressure ($P_{st} / P_{st,d}$)

δ	= Leakage ratio
ϕ	= fan power savings
χ_1	= Pressure loss fraction ($\Delta P_{1,d} / H_d$)
χ_2	= Pressure loss fraction ($\Delta P_{2,d} / H_d$)
χ_3	= Pressure loss fraction ($P_{st,d} / H_d$)

Subscripts

c	= Cooling, cold deck
d	= Design value
h	= Heating, hot deck
v	= VSFS
max	= Maximum
min	= Minimum

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APPENDIX

The procedure of the analysis is demonstrated as followings. The accuracy and resolution of all the sensors were collected from sensor specifications. The calculation of the cold airflow ratio involves multivariable relationship, which requires the uncertainty analysis of error propagation. The error propagation of all the variables results in an uncertainty estimate given by [Figliola and Beasley, 2000]:

$$e = \pm \sqrt{\sum_{i=1}^n (\theta_i \cdot e_{x_i})^2} \quad (A1)$$

$$\text{Where: } \theta_i = \left. \frac{\partial y}{\partial x_i} \right|_{x=\bar{x}} \quad i = 1, 2, \dots, n$$

Errors of static pressure ratio and fan power savings ratio (ordinates of the charts) in Figure (6), Figure (7) and Figure (9) are very small compared to the error ranges of cold airflow ratio, and therefore neglected.